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# A study of the non-wetted flow of water in teflon tubes

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# A STUDY OF THE NON-WETTED FLOW OF WATER IN TEFLON TUBES

by

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Professor in Charge

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## ABSTRACT

Experimental Fanning friction factor results are reported for the non-wetted flow of water in vertical Teflon tubes over the range of Reynolds modulus from about 700 to 50,000. The investigation involved three tubes of 3/8", 3/4", and 1-1/4" nominal inside diameter, each consisting of an inlet section of about fifty diameters length smoothly joined to a test section thirty to forty-eight inches long which could be turned end for end. Results differing by as much as 25% were obtained for flow in opposite directions in the test section, presumably due to imperfect pressure tap holes in the Teflon tubes, but averaged results agree very well with established empirical and theoretical equations for Fanning friction factors in smooth wetted glass and metallic tubes. Results of Atallah in a previous investigation involving horizontal Teflon tubes appear to be incorrect because of failure to correct for the error due to imperfect pressure tap holes. It was concluded that there is no substantial difference in friction factors for wetted or non-wetted flow or for horizontal or vertical flow, at least for the system investigated. Limited experimental results

with air flowing in Teflon tubes strengthen this conclusion.

A shell and single tube heat exchanger, with steam condensing on the outside and water flowing on the inside of a test section of Teflon tubing, was designed and constructed to study heat transfer to water in non-wetted flow. Although no heat transfer data have been obtained, several of the explanations concerning the discrepancies between wetted and non-wetted liquid metal heat transfer have been considered and found to be inadequate when considered independently. While no directly supporting data are available, an additional surface resistance to heat transfer, such as an oxide film which may be removed when wetting occurs, appears to be quite plausible, although non-wetting itself may create a resistance at the non-wetted interface, possibly due to a molecular rearrangement.

## INTRODUCTION

In recent years considerable interest has been aroused in the use of liquid metals as heat transfer media because of their high boiling points, resistance to thermal decomposition, high heat transfer coefficients, and low pumping power requirements for a given system. The liquid metal systems of particular interest, such as mercury flowing in steel tubes, are frequently of such a nature that the liquid metal does not wet the tube wall. Much of the disagreement in experimental heat transfer coefficients for liquid metals is probably due to this phenomenon of non-wetting, which has been shown to create an electrical resistance at the solid-liquid interface (Ref. 8) and which is believed to create a thermal resistance as well.

Although considerable work has been done with heat transfer involving liquid metals, very little work has been concerned directly with the effect of non-wetting on heat transfer and fluid flow. Besides the obvious value of a study of the effect of non-wetting on heat transfer, a study of the effect of non-wetting on friction factors is desirable to allow prediction of pumping power requirements in non-wetted liquid metal systems and in non-wetted plastic pipes, such as polyethylene and Teflon, which are becoming increasingly common. Friction factor data for non-wetted flow might also be useful for predicting heat transfer coefficients for non-wetted flow by means of the analogy between heat and momentum transfer, when this analogy applies.

The purpose of this project was to extend the work on non-wetted flow originated at Lehigh University several years ago (Ref. 1). The system water flowing in Teflon tubes was chosen because of convenience and because water does not wet Teflon (contact angle of 108$^{\circ}$, Ref. 3). In the previous work at Lehigh, friction factors for the horizontal flow of water in the same three Teflon tubes used in this investigation were reported and were found to be considerably higher, at all Reynolds numbers, than published friction factors for wetted flow in glass and metallic tubes. It was also observed that the friction factors tended to approach published values for wetted flow as the diameter of the Teflon tube increased.

Several possible reasons were given for the discrepancies between friction factors for wetted and non-wetted flow. It was concluded that the high values of friction factors under non-wetted conditions were due to entrainment of air bubbles in the liquid stream. It was believed that these air bubbles tended to attach themselves to the upper surface of the tube, thus increasing the roughness and reducing the effective diameter of the tube.

The objectives of this investigation were three-fold. First, it was desired to check the entrained air bubble theory by measuring friction factors in vertical flowing Teflon tubes. It is apparent that if the entrained air bubble theory is correct, the bubbles will tend to be washed away in vertical flow and hence lower friction factors would be expected in vertical flow than in horizontal flow. The friction factor referred to in this investigation is the dimentionless Fanning friction factor, defined by the relationship

$$f = \frac{\pi^2 \rho \ g_c D^5 \Delta P}{32 \ L \ w^2}$$

where $\rho$ = fluid density

$g_c$ = gravitational constant, acceleration due to gravity

$D$ = tube diameter

$\Delta P$ = pressure drop across tube of length L

$L$ = tube length between pressure taps

$w$ = fluid weight rate of flow

The experimental friction factors may be compared with the well established equations for the flow of liquids in smooth wetted circular pipes,

$$f = \frac{16}{Re} \quad \text{for laminar flow}$$

$$f = .00140 + \frac{.125}{Re^{.32}} \quad \text{for turbulent flow (Ref. 7)}$$

where $Re = \frac{DV\rho}{\mu}$ Reynolds dimensionless number

$V$ = linear fluid velocity

$\mu$ = fluid viscosity

A Reynolds number of about 2100 is generally accepted as the lower limit for turbulent flow, although laminar flow may exist at Reynolds numbers considerably higher than 2100 under the proper conditions.

The second objective of this investigation was to measure friction factors for air flowing in Teflon tubes to determine whether results similar to those with water could be obtained. It would seem that the friction factors obtained for air flow in Teflon tubes should be similar to friction factors for air flow in metallic or glass tubes, since the intermolecular forces between air and Teflon, at room temperature, will be very small, as will the forces between air molecules and glass or metal.

The third objective of this investigation was to study the inside water film heat transfer coefficient for water flowing in Teflon tubes with steam condensing on the outside. While Teflon does not promise to be of widespread practical value as a heat transfer material, because of its extremely low thermal conductivity (1.7 BTU / hour $ft^2$ °F / inch), it was believed that a study of heat transfer under non-wetted conditions might shed some light on some of the discrepancies of liquid metal heat transfer. In addition, Teflon tubing may achieve considerable practical importance where extremely corrosive conditions exist, such as in handling hydrofluoric acid.

Much of the literature on liquid metal heat transfer coefficients is in serious disagreement, frequently by 100% or more. As previously mentioned, it seems quite likely that the cause of this disagreement is wetting, a factor difficult to determine in any case but especially difficult in liquid metal heat transfer systems where a fraction of one percent impurity may cause wetting in a system which is believed to be non-wetting.

While the water-Teflon system is very

convenient and also very easy to keep non-wetting, the mechanism of heat transfer in a fluid such as water, with a Prandtl modulus of about 10 when cold, is completely different from the mechanism of heat transfer in a liquid metal, with a Prandtl modulus of about .01. Lyon (Ref. 8) gives a very clear picture of the qualitative effects of the different mechanisms of heat transfer (Figure 1). In the laminar layer near the wall all heat and momentum transfer takes place by molecular motion or, in the case of liquid metals, by electron motion. In the buffer layer heat and momentum are transferred by a combination of molecular and eddy diffusion, and in the turbulent core eddy diffusion is predominant, although in liquid metals molecular diffusion of heat may still be considerable.

Since the Prandtl modulus represents the ratio of molecular diffusivity of momentum to molecular diffusivity of heat, for a given fluid, it can be seen that a fluid with high Prandtl modulus such as water will have poor molecular diffusivity of heat compared with momentum. The temperature distribution curve is therefore different from the velocity distribution in the laminar and buffer layers with very poor heat transfer through the laminar and buffer layers (see Figure 1). In a liquid metal, on the other hand, with a Prandtl modulus of about .01, the high molecular diffusivity will increase heat transfer in the laminar and buffer regions where the velocity distribution discourages heat transfer by eddy diffusion. The assumption has been made throughout this discussion that there is no slip at the wall, which is true for wetted flow.

If the flow is non-wetted, however, the situation may be altered. If the velocity at the wall is not zero, for instance, as might be expected in non-wetted flow because of the relatively greater molecular cohesion than adhesion to the wall, the high resistance to heat transfer in the laminar and buffer regions might be reduced by increased eddy diffusion in these regions. This could account for the higher than expected heat transfer coefficients which a few investigators have obtained in non-wetted flow. An investigation of heat transfer in the Teflon-water system should easily show up any decrease in the laminar and buffer region resistance since these are controlling in this system and any appreciable decrease should increase

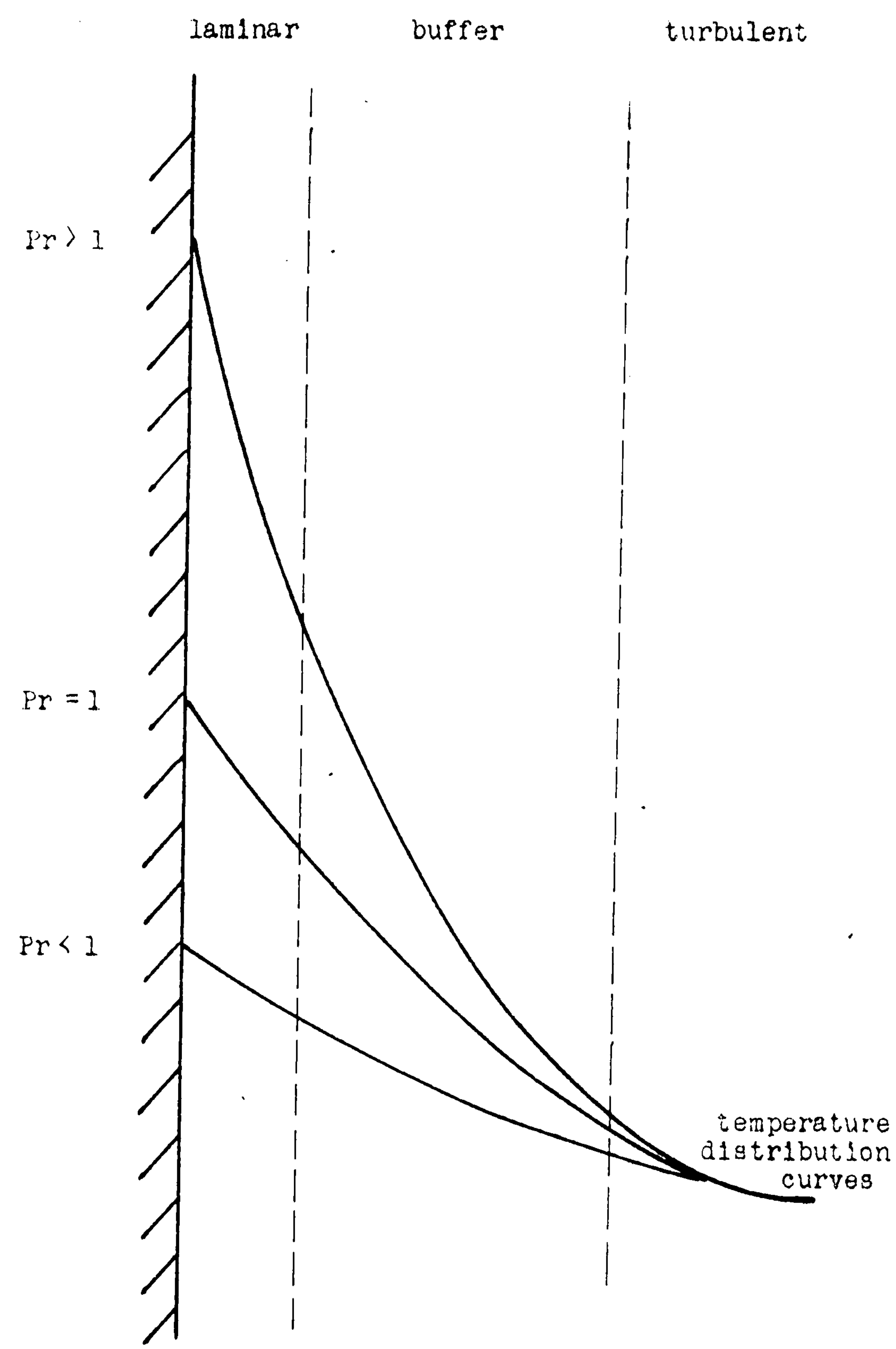



FIGURE 1 --- EFFECT OF PRANDTL NUMBER ON
MECHANISM OF HEAT TRANSFER

the overall coefficient considerably. It may be noted that the high friction factors reported by Atallah for non-wetted flow should also lead to high heat transfer coefficients according to the analogy between heat and momentum transfer.

If there is an added thermal resistance at the non-wetted interface, however, as many investigators have concluded, it would not be expected to show up appreciably in the water-Teflon system because the effect of an additional thermal resistance at the interface will not be as marked in a system where the initial resistance due to the poor heat transfer through the laminar and buffer region is already high.

Strong substantiation of the existance of an additional thermal resistance at the non-wetted interface is given by the conclusive measurement of an interfacial electrical resistance when non-wetting exists. Some of the predominant explanations for this electrical resistance and for the thermal resistance which is believed to be associated with it, are: 1) gas bubble entrainment or insulating gas layer at the interface; 2) distortion of velocity profile; 3) free surface layer of liquid with different properties, such as viscosity, from the bulk of the fluid; 4) oxide film at the solid-liquid interface; and 5) the existance of an interfacial resistance, similar to that supposed to exist at the interface in mass transfer, which may be due to a molecular rearrangement of molecules near the interface because of the unbalanced forces present near the surface.

## DESCRIPTION OF APPARATUS

The apparatus for each of the three phases of work undertaken is described in the following:

### Part I: Measurement of Friction Factors for Water Flowing in Teflon Tubes

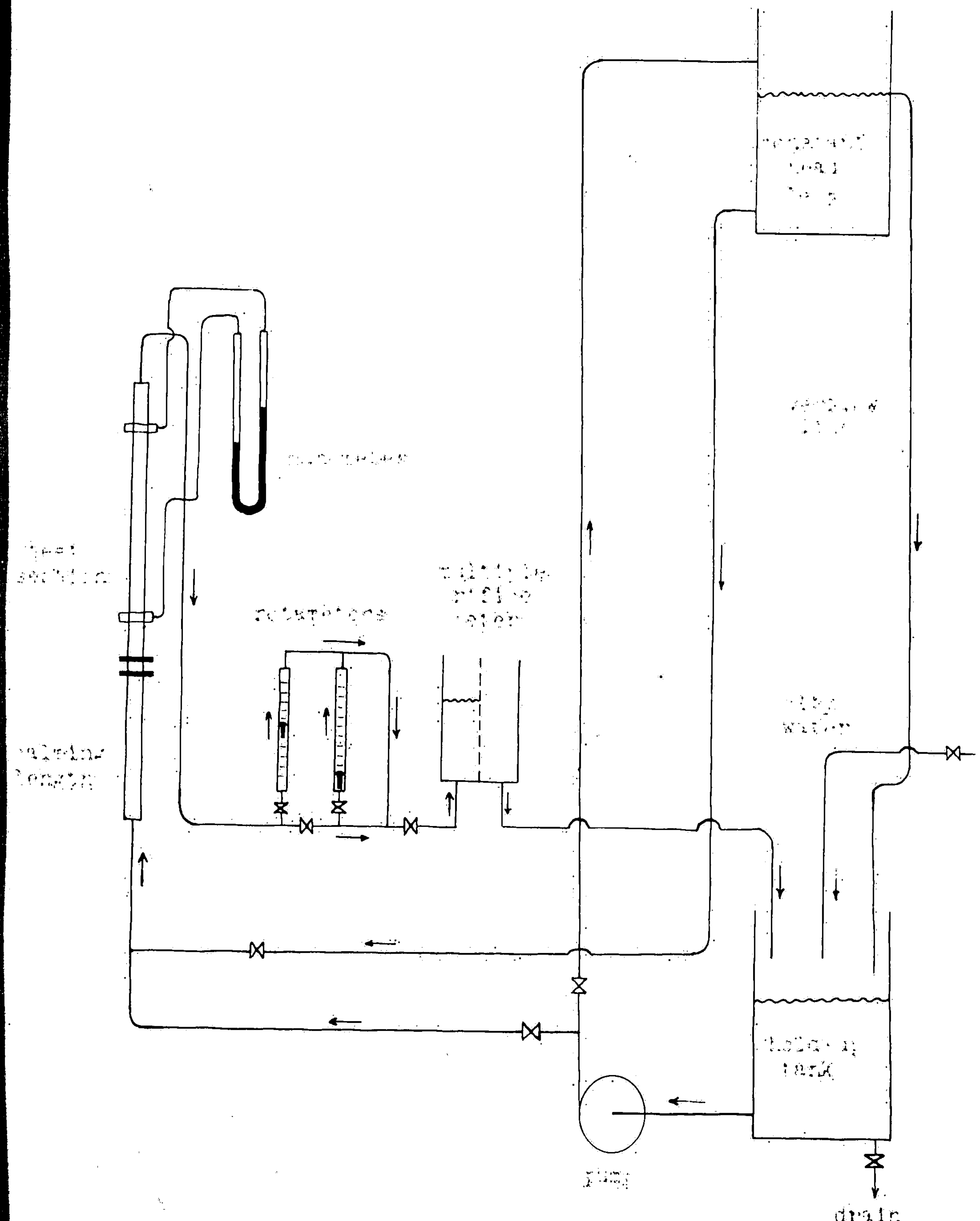
A diagram of the apparatus used in this phase of the work is shown in Figure 2. The apparatus consisted primarily of a hold-up tank on the floor from which water was pumped to a constant head tank at an elevation of about twenty feet. Part of the water to the constant head tank flowed by gravity through the test section and the remainder overflowed back into the hold-up tank. Provision was made to allow water to be pumped directly through the test section in order to obtain flow rates higher than those possible with gravity flow. It was observed that the centrifugal pump discharge pressure was steady at all flow rates, which minimized the effect of pulsation when water was pumped directly through the test section.

The temperature of the water in the system was controlled by continuously draining some of the water from the hold-up tank and replacing it with cold city water.

The three Teflon tubes investigated were the

FIGURE ([illegible])

APPARATUS FOR MEASUREMENT OF FRICTION FACTORS
FOR WATER FLOW IN TEFLON TUBES



same tubes used in the previous work, except that the pressure tap holes on the 3/8" tube were redrilled closer together because there was some distortion of the tube at the original pressure taps. The pressure taps in the other two tubes were carefully cleaned and filed lightly to remove burrs which might create undue turbulence. In addition, each of the tubes was cut at the end of the calming section and fitted with clamps which fit into slots machined in the tube circumference to hold the two sections together. The ends of the two sections at the joint were machined square so that they fit tightly and did not leave a rough edge on the inside which might create undue turbulence, and a machined steel sleeve fit over the joint for rigidity. A photograph of the 3/4" Teflon tube, showing the method of joining and a pressure tap, is shown in Figure 3. The purpose of having the tubes in two sections, a calming section of about 50 diameters length and a test section, was to allow the test section to be reversed so that pressure drops could be measured for flow in opposite directions to attempt to compensate for errors which might arise from burrs, dents, or lack of perfect symmetry at the pressure taps.

The dimensions of all three tubes are tabulated in Table (I), including the ratio of calming length to I.D.

Flow through the tubes was controlled by means of a needle valve at the outlet, permitting very fine control. Since the tubes were vertical, they were obviously flowing full at all times. The flow rates were measured by means of two Fisher and Porter rotameters, one calibrated from 0.4 to 3.4 lbs./min. and the other calibrated from 3.6 to 11.0 lbs./min., and a specially constructed multiple orifice meter calibrated from 10 to 150 lbs./min. Calibration curves for the three flow meters are included in the appendix (Figures 9,10,11) together with a sketch of the multiple orifice meter. Calibration was done by weight rate of flow.

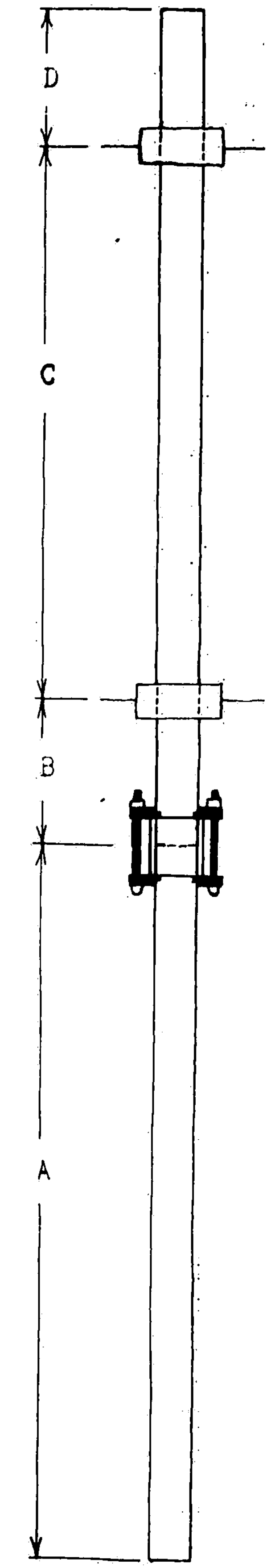



TABLE I

DIMENSIONS OF TEFLON TUBES
( in inches)

| | | | |
|---|---|---|---|
| Nominal I.D. | 3/8 | 3/4 | 1-1/4 |
| Nominal O.D. | 1/2 | 1 | 1-11/16 |
| Actual I.D. | 0.372 | 0.721 | 1.228 |
| Actual O.D. | 0.516 | 1.032 | 1.723 |
| A, calming length | 15-1/2 | 27-1/2 | 53-1/2 |
| B | 9-1/2 | 6-1/4 | 6 |
| C, tested length | 38 | 32 | 30 |
| D | 9-1/2 | 6-1/4 | 6 |
| $\frac{A\;\;B}{I.D.}$ | 67.2 | 46.8 | 48.5 |

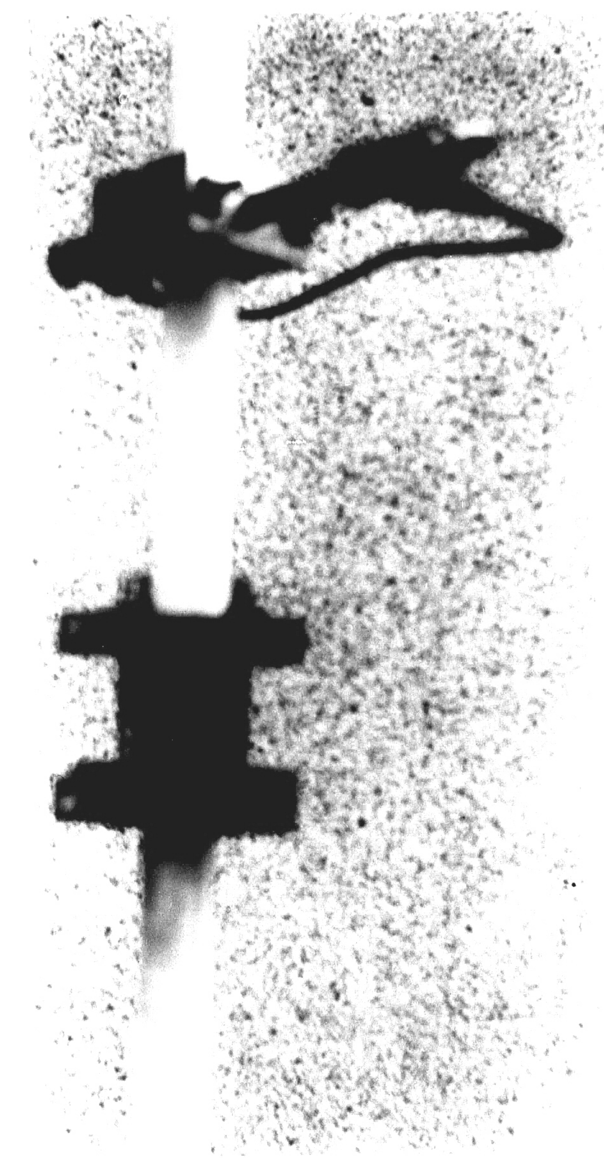


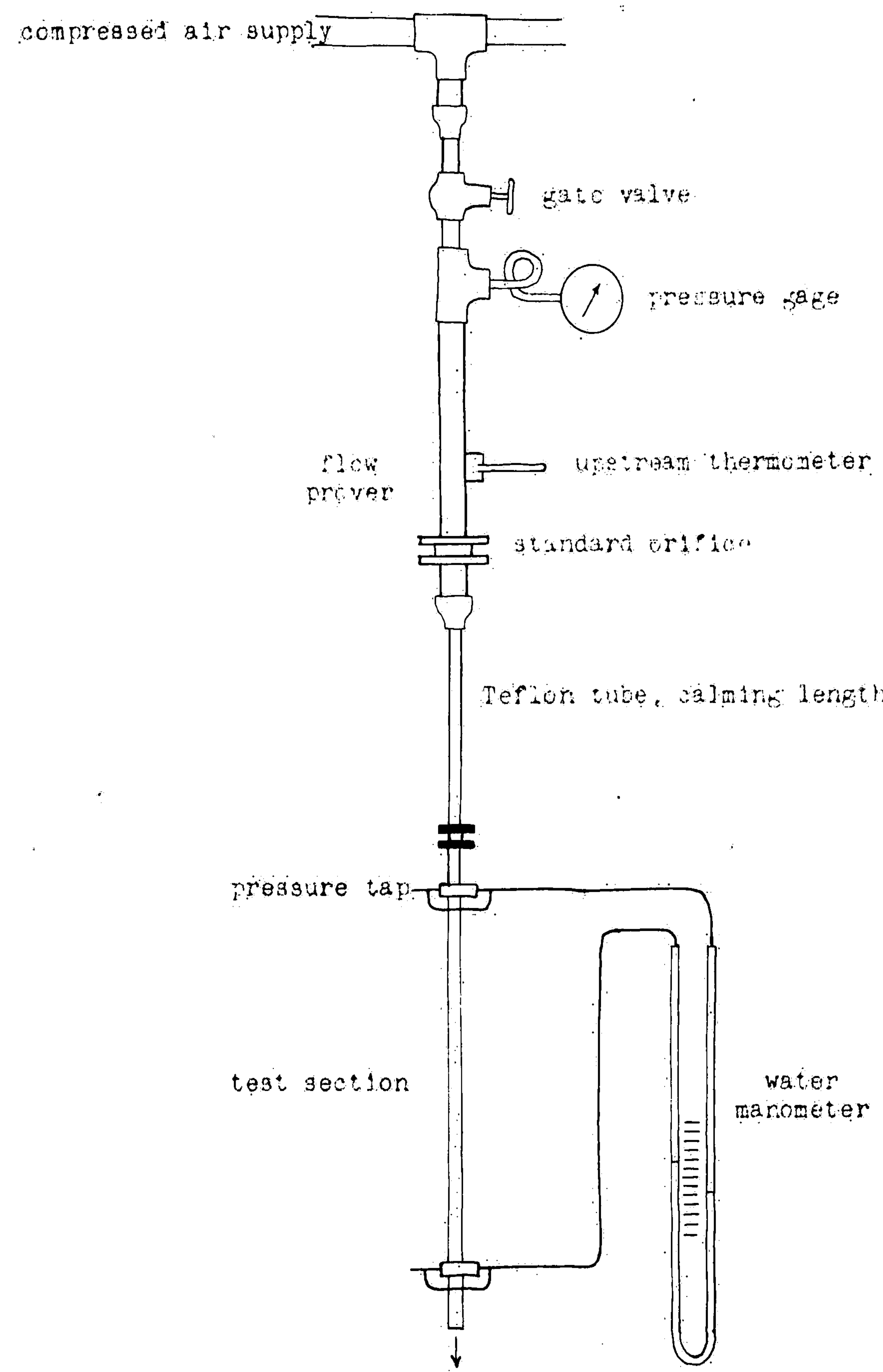
FIGURE 3 ---- PHOTOGRAPH OF 3/4" TEFLON TUBE, SHOWING METHOD OF JOINING AND A PRESSURE TAP

Part II: Measurement of Friction Factors for Air Flowing in Teflon Tubes

A diagram of the apparatus used in this phase of the work is shown in Figure (4). The same 3/4" nominal I.D. Teflon tube and pressure taps used in Part I were used in Part II. A supply of compressed air at about 50 psig was available and was passed through an American Meter Company critical flow prover. By throttling the compressed air to various upstream pressures to the flow prover, varying flow rates could be measured through the orifice. The tube to be tested was connected tightly with the outlet of the flow prover, and the pressure drop through the test section was measured by means of a water manometer.

FIGURE (4)

APPARATUS FOR STUDY OF FRICTION FACTORS FOR FLOW OF AIR IN TEFLON TUBES



Part III: Measurement of Overall Heat Transfer Coefficients for Water Flowing in Teflon Tubes with Steam Condensing on Outside.

The apparatus constructed for this phase of the work consists primarily of a heat exchanger which is to be used with the same piping arrangement used in Part I, including pump, hold-up and constant head tanks, and flow meters (see Figure 2). Details of the heat exchanger itself are shown in Figure 5.

The shell of the heat exchanger consists of an 8'-0" length of 6" standard pipe, flanged at each end, with two 4"x8" port holes cut into the pipe wall at one end to permit observation of the condensate on the Teflon tube. Steam, available at high pressure, is passed through a pressure regulating valve which can be used to regulate the pressure, and hence temperature, of the steam entering the shell. The steam is introduced at four points along the length of the shell in order to secure proper distribution.

The heat exchanger is mounted at a slight angle so that any steam which condenses in the outer 6" diameter shell can be collected at a tap

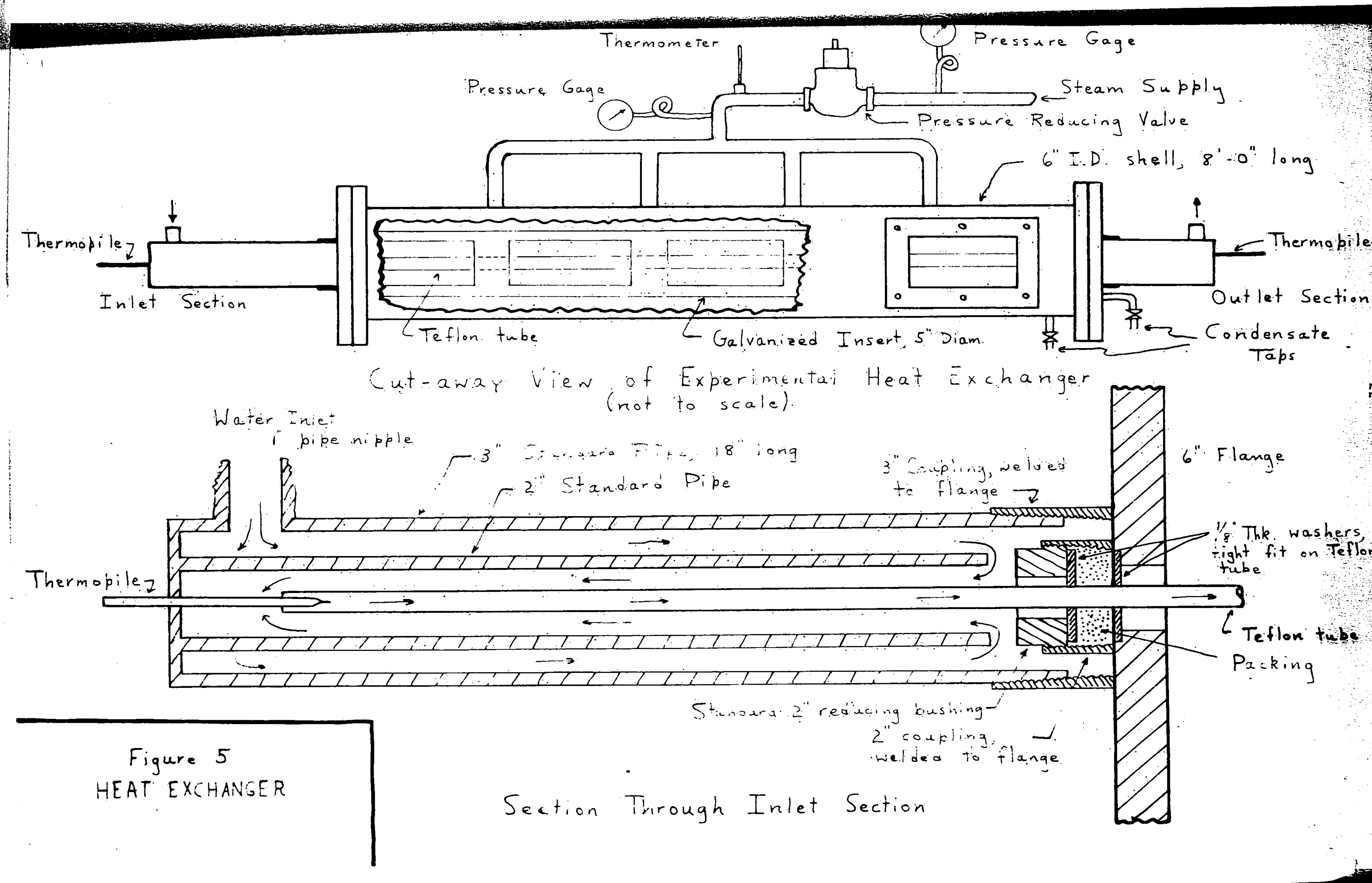



Figure 5
HEAT EXCHANGER

at one end. The steam which condenses on the Teflon tube, however, is collected separately by a 5" diameter galvanized insert held inside and concentric to the 6" shell. This insert acts as a steam diffuser, tube support,and condensate collector. Condensate from this insert is removed by a separate tap in the flange at the low end.

A 10'-0" length of special thin walled Teflon tube, 13/32" I.D., 30 mils wall thickness, is held concentric to both 6" shell and 5" diameter insert. Eighteen inches of the tube extend into the water inlet chamber where it acts as a calming section. The water inlet chamber, shown in detail in Figure (5), acts as a mixing box to enable measurement of the bulk inlet temperature, and acts as a stuffing box to prevent steam leaking into the water. In the event that any slight amount of steam should leak past the packing, it will enter the water stream before measurement of the bulk temperature which will not cause any error. The exit pipe arrangement serves the dual purpose of mixing box and stuffing box at the downstream end.

## EXPERIMENTAL PROCEDURE

### Part I: Measurement of Friction Factors for Water Flowing in Teflon Tubes

The hold-up tank was filled with water and water was pumped up to the constant head tank to maintain a constant head of about twenty feet. Water was drained from the hold-up tank and replaced with cold city water at a rate to maintain the water in the system at about $49^{o}F$, within one degree. At times the water temperature could not be maintained at $49^{o}F$, so it was maintained at $53^{o}F$, within one degree.

When the water temperature became constant the needle valve at the outlet of the tube being tested was opened to the desired flow rate. Care was taken to make certain that all air which might be trapped in high points in the piping were flushed out, especially for the first run in any day. After allowing several minutes for the system to come to equilibrium, as evidenced by constant readings of rotameters and orifice meter, the manometer taps were carefully bled to make sure that all air was removed from the lines. Occasionally this required vigorous flushing. Manometer and flow rate readings were then recorded. The manometer taps were bled a second time to make sure the reading was steady. If the reading had changed, the new reading was also recorded and the procedure repeated until a reproducible reading was obtained for a given flow rate. Very little trouble was experienced with the manometers except at very low readings (less than 1/2" reading on the carbon tetrachloride over water manometer), where the manometer reading occasionally varied as much as 0.05 inches, never reading constant. In these cases an average value was estimated. It is probable that the difficulty in obtaining some of these readings was due to flow being in the transition region where it is conceivable that the pressure drop could fluctuate.

Some of the higher flow rates were obtained by pumping water from the hold-up tank directly through the tested section instead of using gravity flow. The head of the pump was fifty feet at 20 gpm which gave a quite high flow rate, and the pump output pressure was observed to be very steady. No differences were observed between results obtained using the pump or the constant head tank.

Some of the first data were taken using a Niagara Meter Company nutating disc water meter to measure the water flow rate. It was observed that this meter caused a pulsating flow, due to water filling and emptying the meter chambers, and these data weren't used in the results. The nutating disc meter was replaced by a specially constructed multiple-orifice meter, a sketch of which appears in the appendix (Figure 11), which allowed reasonably precise flow rate measurements between 10 and 120 lbs./min.

Each tube was tested with flow in both directions. In reversing the tube, the pressure taps were left on but the manometer connections were removed. The test section was then reversed, leaving the calming section in place. The test section was thoroughly cleaned with strong soap and a rather stiff brush each time a tube was reversed or a different diameter tube was tested. No indication of wetting of the test section was ever apparent. Although Teflon is relatively soft, it is not believed that the cleaning brush affected the friction characteristics of the tubes in any way and, if it did scratch, the scratches would be longitudinal which would minimize their effect on the friction factor.

A few runs were made with the 3/8" and 3/4" Teflon tubes in a horizontal position in order to see if the results of the previous investigation could be reproduced. The procedure was essentially the same as for vertical flow, except that less difficulty was encountered in flushing out air trapped in the high points of the piping.

All of the readings, in this and subsequent phases of the project, were taken at random.

Part II: Measurement of Friction Factors for Air Flowing in Teflon Tubes.

All of the data obtained for air flowing in Teflon tubes were obtained using the 3/4" Teflon tube. After assembling the flow prover with the proper size standard orifice, the Teflon tube was attached at the outlet. The compressed air gate valve was opened to give the desired pressure upstream from the standard orifice, and the air was allowed to flow until both the upstream and downstream temperatures became constant. About twenty minutes were required to attain equilibrium. The upstream pressure and temperature (measured at the Teflon tube outlet), and the manometer reading were then recorded. It was assumed that the air flowing in the tested tube was at atmospheric pressure and the outlet temperature.

Part III: Measurement of Overall Heat Transfer Coefficients for Water Flowing in Teflon Tubes with Steam Condensing on the Outside.

The steam pressure regulating valve must be adjusted for the required downstream pressure, as determined from the temperature desired in the heat exchanger. The water flow rate must then be adjusted to the desired flow rate.

After equilibrium has been reached in the system, as determined by constant water inlet and outlet temperatures and constant steam temperature and pressure in the heat exchanger, the condensate in the inner insert and the outer shell is drained through the appropriate stopcocks. The measurement of time is then started, and the following information recorded: 1) upstream steam pressure; 2) downstream steam pressure and temperature; 3) shell steam pressure and temperature; 4) inlet water temperature, as determined by a thermometer within one degree; 5) precise water temperature difference between inlet to Teflon tube and outlet, as measured by thermopile; 6) water flow rate. After a measured period of time the condensate collected in the inner insert, from the steam condensed on the test section, is drained and measured, taking care to avoid losses by flashing.

Particular attention must be given to the following points: 1) leaks of water or steam through the packing boxes must be avoided; 2) leakage of condensate from the inner insert into the shell must be avoided; and 3) both the inside and outside of the tested section must be carefully cleaned periodically to ensure non-wetting.

TABLE II

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/8" nominal I.D. with horizontal water flow, end 1 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 1 | 1.45 | 0.70 | 1085 | 0.0168 |
| 2 | 1.95 | 1.00 | 1480 | 0.0129 |
| 3 | 3.29 | 3.20 | 2490 | 0.0146 |
| 4 | 10.00 | 23.50 | 7570 | 0.0126 |
| 5 | 4.96 | 7.00 | 3760 | 0.0141 |
| 6 | 2.45 | 1.80 | 1855 | 0.0148 |
| 7* | | | | |
| 8* | | | | |
| 9* | | | | |
| 10 | 1.29 | 0.70 | 980 | 0.0207 |
| 11 | 6.38 | 10.55 | 4830 | 0.0128 |
| 12 | 9.00 | 19.65 | 6820 | 0.0120 |
| 13 | 2.89 | 2.40 | 2190 | 0.0142 |

* This run was not calculated because flow was measured with the nutating disc water meter.

TABLE III

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/8" nominal I.D. with horizontal water flow, end 2 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 14 | 3.08 | 2.50 | 2330 | 0.0130 |
| 15 | 7.12 | 11.30 | 5400 | 0.0110 |
| 16* | | | | |
| 17* | | | | |
| 18 | 14.17 | 11.65** | 10700 | 0.0096 |
| 19* | | | | |
| 20 | 10.88 | 24.80 | 8240 | 0.0104 |

* This run was not calculated because flow was measured with the nutating disc water meter.

** This pressure drop is in inches of red oil, of 2.95 specific gravity, under water.

## TABLE IV

## CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/8" nominal I.D. with vertical water flow, end 1 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 151 | 5.50 | 6.20 | 4160 | 0.0128 |
| 152 | 4.46 | 4.35 | 3290 | 0.0147 |
| 153 | 5.80 | 6.78 | 4380 | 0.0126 |
| 154 | 3.55 | 3.00 | 2680 | 0.0148 |
| 155 | 3.27 | 1.55 | 2470 | 0.00903 |
| 156 | 2.85 | 1.07 | 2150 | 0.00822 |
| 157 | 3.24 | 1.50 | 2450 | 0.00892 |
| 158 | 4.40 | 4.84 | 3330 | 0.0156 |
| 159 | 3.88 | 3.25 | 3900 | 0.0135 |
| 160 | 4.10 | 3.60 | 3100 | 0.0133 |
| 161 | 8.40 | 12.10 | 6350 | 0.0107 |
| 162 | 3.13 | 1.22 | 2370 | 0.00777 |
| 163 | 2.50 | 0.86 | 1890 | 0.00858 |
| 164 | 3.32 | 1.52 | 2510 | 0.00860 |
| 165 | 10.40 | 17.07 | 7870 | 0.00985 |
| 166 | 14.03 | 27.15 | 10600 | 0.00860 |
| 167 | 12.75 | 23.56 | 9650 | 0.00905 |
| 168 | 8.95 | 13.84 | 6770 | 0.0108 |
| 169 | 3.63 | 3.02 | 2740 | 0.0143 |
| 170 | 4.34 | 3.78 | 3280 | 0.0125 |
| 176 | 0.85 | 0.22 | 643 | 0.0190 |
| 177 | 3.12 | 1.40 | 2360 | 0.00897 |
| 178 | 2.32 | 0.81 | 1756 | 0.00940 |
| 179 | 1.56 | 0.51 | 1180 | 0.0131 |
| 180 | 2.72 | 1.02 | 2060 | 0.0086 |
| 181 | 14.10 | 31.25 | 10680 | 0.00981 |
| 182 | 21.50 | 18.54* | 16250 | 0.00820 |
| 183 | 25.00 | 24.00* | 18900 | 0.00785 |
| 184 | 18.30 | 13.40* | 13830 | 0.00815 |
| 185 | 13.90 | 28.72 | 10500 | 0.00930 |
| 186 | 24.90 | 23.95* | 18850 | 0.00786 |
| 187 | 3.05 | 2.15 | 2310 | 0.0144 |
| 188 | 1.20 | 0.46 | 907 | 0.0199 |
| 189 | 1.70 | 0.63 | 1287 | 0.0136 |

* This pressure drop is in inches of red oil, of 2.95 specific gravity, under water.

TABLE V

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/8" nominal I.D. with vertical water flow, end 2 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 135 | 1.25 | 0.68 | 943 | 0.0274 |
| 136 | 2.94 | 2.02 | 2225 | 0.0145 |
| 137 | 8.07 | 10.90 | 6100 | 0.0105 |
| 138 | 0.45 | 0.24 | 340 | 0.0738 |
| 139 | 3.18 | 2.15 | 2410 | 0.0132 |
| 140 | 10.19 | 16.32 | 7700 | 0.00983 |
| 141 | 7.48 | 9.63 | 5670 | 0.0107 |
| 142 | 2.93 | 1.90 | 2220 | 0.0138 |
| 143 | 0.99 | 0.42 | 749 | 0.0267 |
| 144 | 5.60 | 5.89 | 4240 | 0.0117 |
| 145 | 9.80 | 15.09 | 7410 | 0.00982 |
| 146 | 7.55 | 9.67 | 5710 | 0.0106 |
| 147 | 2.10 | 1.09 | 1590 | 0.0154 |
| 148 | 1.31 | 0.54 | 992 | 0.0197 |
| 149 | 10.56 | 16.95 | 8000 | 0.00948 |
| 150 | 8.12 | 11.10 | 6140 | 0.0105 |
| 192 | 2.90 | 1.85 | 2190 | 0.0137 |
| 193 | 1.42 | 0.50 | 1075 | 0.0155 |
| 194 | 3.14 | 2.00 | 2380 | 0.0126 |
| 195 | 8.05 | 10.62 | 6090 | 0.0102 |
| 196 | 5.05 | 4.40 | 3820 | 0.0107 |
| 197 | 12.75 | 23.10 | 9650 | 0.00887 |
| 198 | 12.88 | 23.81 | 9740 | 0.00898 |
| 199 | 14.02 | 27.50 | 10600 | 0.00871 |
| 200 | 26.00 | 23.70* | 19700 | 0.00717 |
| 201 | 22.30 | 17.92* | 16900 | 0.00738 |
| 202 | 23.50 | 17.80* | 16800 | 0.00747 |
| 203 | 10.28 | 15.70 | 7770 | 0.00928 |
| 204 | 2.10 | 0.98 | 1590 | 0.0139 |

TABLE V (CONTINUED)

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 205 | 3.15 | 2.13 | 2380 | 0.0134 |
| 207 | 6.94 | 8.04 | 5250 | 0.0104 |
| 206 | 5.85 | 6.27 | 4430 | 0.0114 |
| 208 | 1.24 | 0.50 | 938 | 0.0203 |
| 209 | 1.78 | 0.72 | 1346 | 0.0142 |
| 210 | 2.63 | 1.40 | 1990 | 0.0126 |

* This pressure drop is in inches of red oil, of 2.95 specific gravity, under water.

FIGURE (6)

FRICTION FACTOR versus REYNOLDS NUMBER
FOR TEFLON TUBE OF 3/8" NOMINAL I. D.
COMPARED WITH PUBLISHED DATA

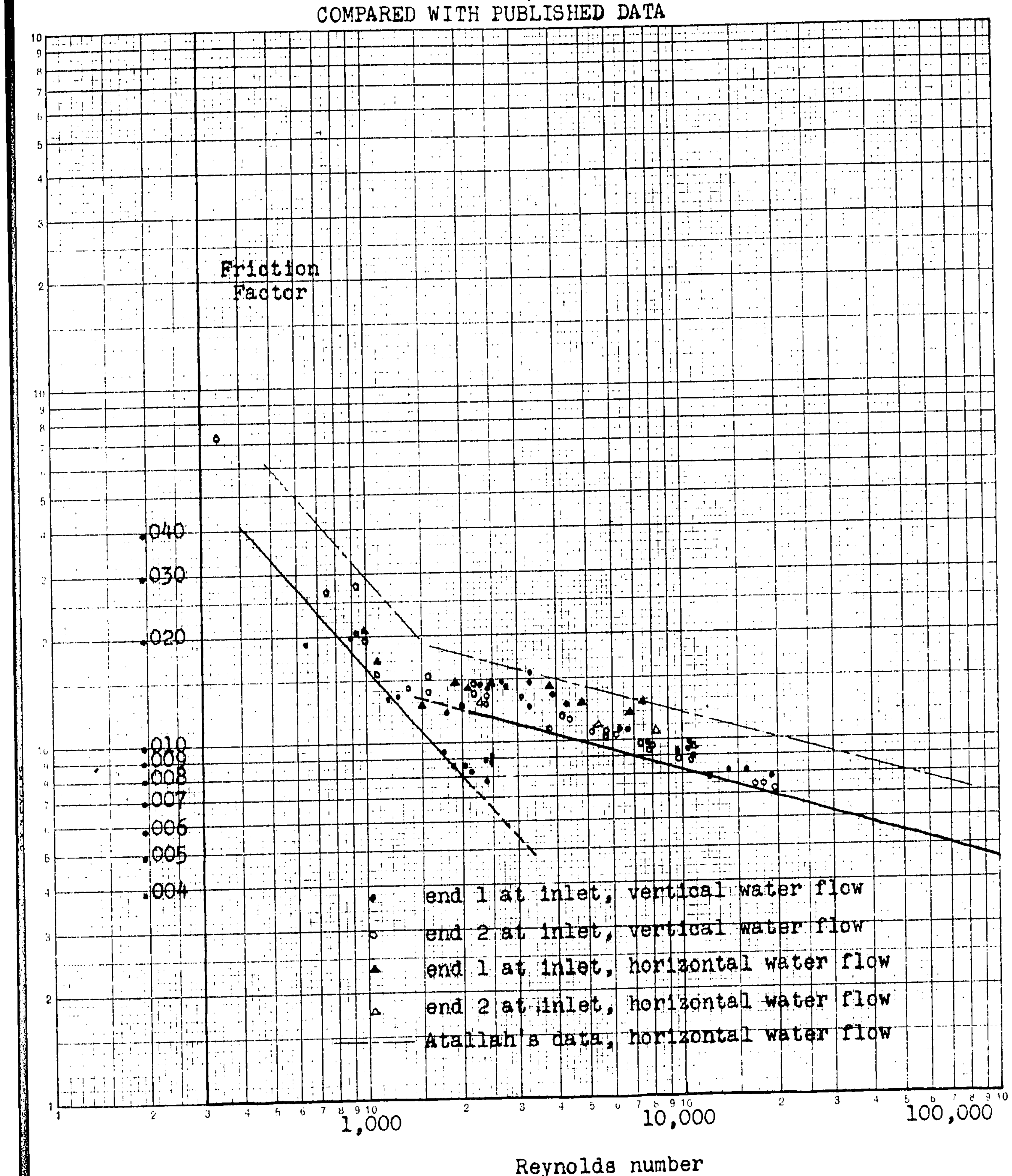


TABLE VI

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/4" nominal I.D. with horizontal water flow, end 1 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 41 | 74.50 | 14.80 | 29100 | 0.00533 |
| 42 | 46.20 | 6.55 | 18100 | 0.00616 |
| 43 | 22.60 | 1.95 | 8690 | 0.00765 |
| 44 | 10.71 | 0.68 | 4190 | 0.0118 |
| 45 | 29.00 | 2.90 | 11340 | 0.00690 |
| 46 | 51.50 | 7.80 | 21500 | 0.00590 |
| 47 | 58.60 | 9.80 | 22900 | 0.00572 |
| 48 | 64.30 | 11.25 | 25200 | 0.00544 |
| 49 | 70.30 | 13.30 | 27500 | 0.00538 |
| 50 | 74.80 | 14.75 | 29300 | 0.00528 |
| 51 | 125.80 | 36.20 | 48200 | 0.00479 |
| 52 | 122.80 | 36.00 | 48000 | 0.00481 |
| 53 | 105.00 | 28.50 | 41100 | 0.00518 |
| 54 | 114.50 | 33.00 | 44800 | 0.00505 |
| 55 | 9.68 | 0.65 | 3785 | 0.0139 |
| 56 | 4.92 | 0.25 | 1925 | 0.0207 |
| 57 | 10.93 | 0.60 | 4280 | 0.0101 |
| 58 | 9.88 | 0.50 | 3860 | 0.0102 |

TABLE VII

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/4" nominal I.D. with vertical water flow, end 1 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 59 | 21.30 | 1.80 | 8320 | 0.00795 |
| 60 | 34.50 | 4.02 | 13450 | 0.00677 |
| 61 | 53.20 | 8.30 | 20800 | 0.00587 |
| 62 | 65.20 | 12.00 | 25500 | 0.00565 |
| 63 | 54.30 | 8.90 | 21200 | 0.00604 |
| 64 | 39.80 | 5.30 | 15550 | 0.00670 |
| 65 | 9.45 | 0.50 | 3690 | 0.01122 |
| 66 | 32.30 | 3.60 | 12600 | 0.00692 |
| 67 | 47.80 | 7.08 | 18700 | 0.00620 |
| 68 | 70.30 | 13.47 | 27500 | 0.00546 |
| 69 | 29.60 | 3.07 | 11570 | 0.00701 |
| 70 | 7.48 | 0.32 | 2920 | 0.01145 |
| 71 | 31.20 | 3.53 | 12200 | 0.00726 |
| 72 | 70.40 | 13.45 | 27500 | 0.00543 |
| 73 | 63.30 | 11.25 | 24800 | 0.00562 |
| 74 | 56.00 | 9.43 | 21900 | 0.00603 |
| 75 | 38.20 | 4.80 | 14900 | 0.00659 |
| 76 | 25.05 | 2.40 | 9800 | 0.00768 |
| 77 | 50.60 | 7.60 | 19800 | 0.00600 |
| 78 | 60.40 | 10.57 | 23600 | 0.00588 |
| 79 | 45.30 | 6.56 | 17700 | 0.00648 |
| 80 | 29.00 | 3.10 | 11350 | 0.00748 |
| 81 | 70.30 | 13.58 | 27500 | 0.00557 |
| 82 | 13.40 | 0.48 | 5250 | 0.0104 |
| 83 | 30.00 | 3.20 | 11740 | 0.00722 |
| 84 | 53.00 | 8.51 | 20700 | 0.00614 |
| 85 | 34.80 | 4.20 | 13600 | 0.00705 |
| 86 | 38.10 | 4.89 | 14900 | 0.00683 |

TABLE VII (CONTINUED)

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 87 | 88.40 | 20.15 | 34500 | 0.00524 |
| 88 | 72.60 | 14.51 | 28400 | 0.00558 |
| 89 | 119.10 | 34.51 | 46600 | 0.00494 |
| 90 | 108.50 | 28.62 | 42500 | 0.00488 |
| 91 | 72.80 | 14.75 | 28400 | 0.00563 |
| 92 | 92.60 | 22.27 | 36200 | 0.00527 |
| 93 | 65.90 | 12.24 | 25800 | 0.00573 |
| 94 | 85.70 | 19.43 | 33500 | 0.00531 |
| 95 | 119.10 | 34.30 | 46600 | 0.00484 |
| 96 | 2.17 | 0.07 | 850 | 0.0298 |
| 97 | 9.90 | 0.50 | 3870 | 0.0102 |
| 98 | 7.15 | 0.37 | 2800 | 0.0145 |
| 99 | 3.10 | 0.19 | 1215 | 0,0396 |
| 100 | 5.60 | 0.27 | 2190 | 0.0172 |
| 101 | 9.65 | 0.56 | 3770 | 0.0121 |
| 102 | 10.73 | 0.67 | 4200 | 0.0117 |
| 103 | 7.76 | 0.38 | 3040 | 0.0127 |
| 104 | 3.30 | 0.16 | 1290 | 0.0294 |
| 105 | 2.60 | 0.14 | 1020 | 0.0415 |
| 106 | 22.30 | 2.05 | 8700 | 0.00838 |
| 107 | 13.48 | 0.90 | 5260 | 0.0101 |
| 108 | 19.70 | 1.54 | 7700 | 0.00805 |
| 109 | 15.64 | 1.08 | 6120 | 0.00895 |
| 110 | 5.60 | 0.15 | 2190 | 0.00970 |
| 111 | 9.80 | 0.51 | 3830 | 0.0108 |
| 112 | 20.05 | 1.66 | 8000 | 0.00802 |
| 113 | 17.42 | 1.35 | 6810 | 0.00901 |
| 114 | 18.86 | 1.39 | 7370 | 0.00794 |
| 115 | 8.75 | 0.45 | 3420 | 0.0112 |
| 116 | 18.30 | 1.46 | 7150 | 0.00887 |
| 117 | 30.00 | 3.10 | 11730 | 0.00700 |
| 118 | 48.80 | 7.28 | 19100 | 0.00620 |
| 119 | 65.40 | 11.94 | 25600 | 0.00567 |
| 120 | 58.70 | 10.28 | 22950 | 0.00606 |
| 121 | 18.30 | 1.40 | 6500 | 0.00837 |
| 122 | 11.20 | 0.64 | 3980 | 0.0102 |
| 123 | 48.00 | 7.10 | 18800 | 0.00617 |

TABLE VIII

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 3/4" nominal I.D. with vertical water flow, end 2 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 124 | 9.30 | 0.46 | 3640 | 0.0107 |
| 125 | 34.45 | 5.26 | 13480 | 0.00890 |
| 126 | 53.20 | 11.00 | 20800 | 0.00778 |
| 127 | 29.80 | 3.93 | 11650 | 0.00898 |
| 128 | 69.50 | 17.74 | 27200 | 0.00744 |
| 129 | 52.70 | 10.88 | 20600 | 0.00795 |
| 130 | 41.00 | 7.15 | 16050 | 0.00863 |
| 131 | 34.20 | 5.13 | 13400 | 0.00891 |
| 132 | 10.30 | 0.64 | 4030 | 0.0121 |
| 133 | 45.50 | 8.65 | 17800 | 0.00848 |
| 134 | 31.70 | 4.75 | 12400 | 0.00945 |

TABLE IX

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of water for the Teflon tube of 3/4" nominal I.D. with vertical air flow, end 2 at inlet.

| Run number | Flow rate lbs./sec. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 171 | 0.01153 | 1.00 | 20150 | 0.00826 |
| 172 | 0.01483 | 1.52 | 25900 | 0.00760 |
| 173 | 0.00955 | 0.68 | 16700 | 0.00820 |
| 174 | 0.01285 | 1.20 | 22500 | 0.00800 |
| 175 | 0.01635 | 1.82 | 28600 | 0.00748 |

FIGURE (7)

FRICTION FACTOR versus REYNOLDS NUMBER
FOR TEFLON TUBE OF 3/4" NOMINAL I. D.
COMPARED WITH PUBLISHED DATA

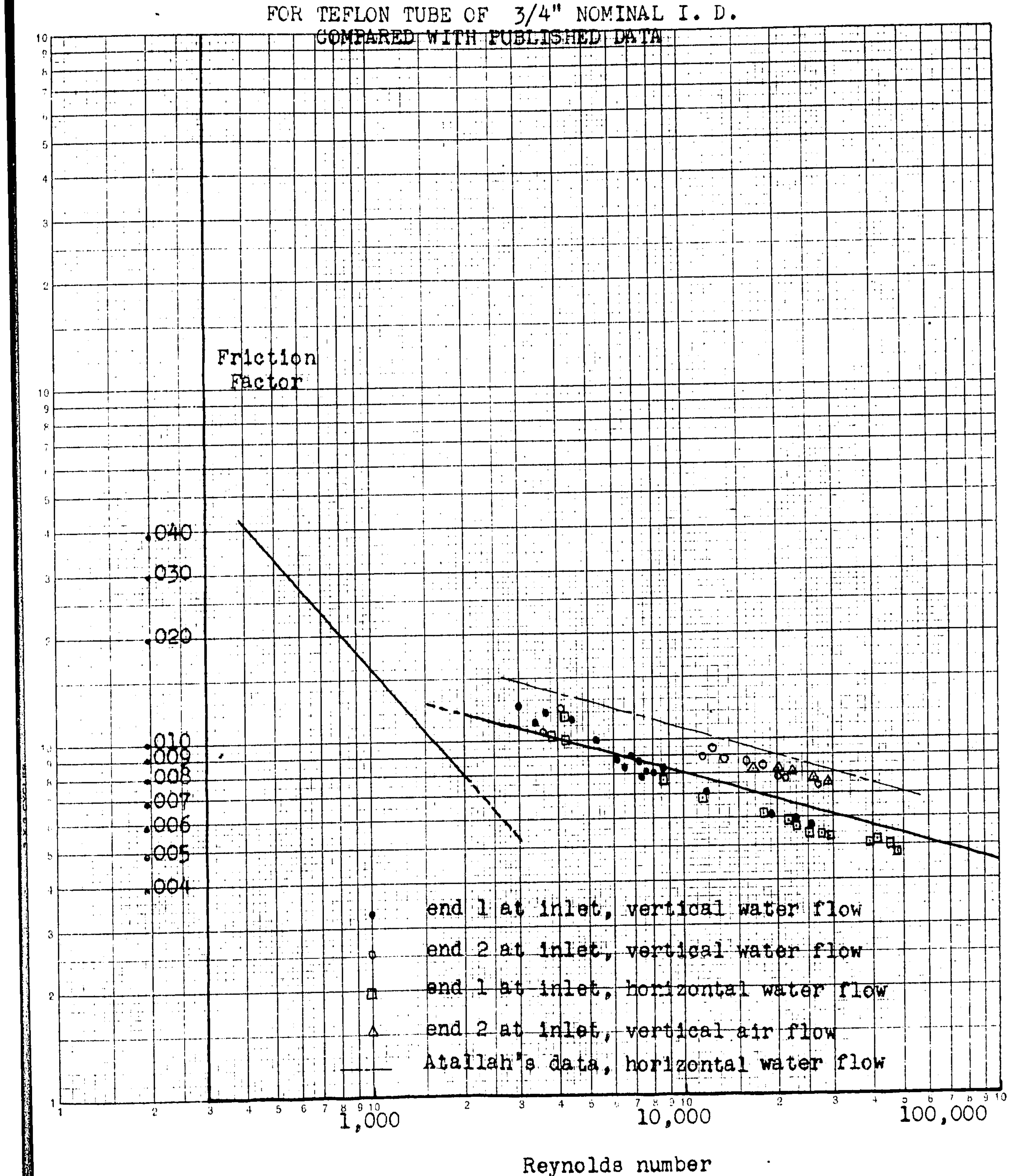


TABLE X

CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 1-1/4" nominal I.D. with vertical water flow, end 1 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 211 | 143.8 | 3.84 | 34000 | 0.00572 |
| 212 | 61.0 | 0.88 | 13970 | 0.00730 |
| 213 | 92.2 | 1.70 | 21100 | 0.00622 |
| 214 | 39.2 | 0.46 | 9000 | 0.00924 |
| 215 | 126.6 | 2.91 | 29000 | 0.00564 |
| 216 | 101.0 | 2.00 | 23150 | 0.00606 |
| 217 | 73.2 | 1.18 | 16750 | 0.00682 |
| 218 | 110.0 | 2.30 | 25200 | 0.00587 |
| 219 | 79.3 | 1.38 | 18180 | 0.00678 |
| 220 | 143.8 | 3.80 | 32900 | 0.00568 |
| 221 | 143.8 | 3.95 | 32950 | 0.00590 |
| 222 | 120.5 | 2.65 | 27600 | 0.00565 |
| 223 | 84.5 | 1.48 | 19360 | 0.00641 |
| 224 | 105.0 | 2.13 | 24100 | 0.00597 |

TABLE XI

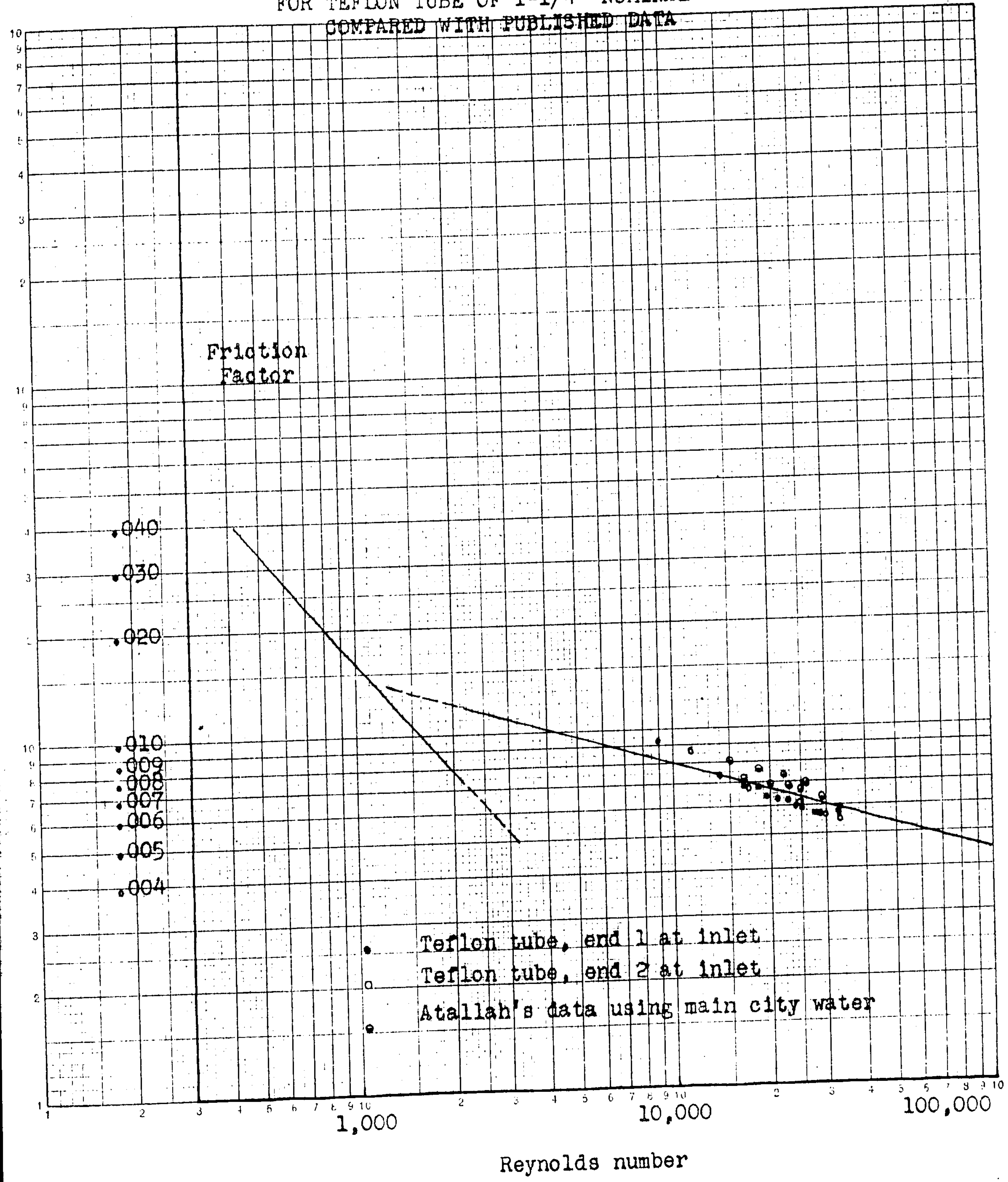
CALCULATED RESULTS

The following are the calculated values for the Reynolds number and the friction factor with the corresponding values of flow rate and pressure drop in inches of carbon tetrachloride under water for the Teflon tube of 1-1/4" nominal I.D. with vertical water flow, end 2 at inlet.

| Run number | Flow rate lbs./min. | Pressure drop in inches | Reynolds number | Friction factor |
|---|---|---|---|---|
| 225 | 143.8 | 3.72 | 32950 | 0.00556 |
| 226 | 75.0 | 1.25 | 17200 | 0.00687 |
| 227 | 109.6 | 2.33 | 25100 | 0.00600 |
| 228 | 50.0 | 0.70 | 11460 | 0.00867 |
| 229 | 81.2 | 1.46 | 18600 | 0.00683 |
| 230 | 128.8 | 3.00 | 29500 | 0.00560 |

FIGURE (8)

FRICTION FACTOR versus REYNOLDS NUMBER
FOR TEFLON TUBE OF 1-1/4" NOMINAL I.D.
COMPARED WITH PUBLISHED DATA



## DISCUSSION OF RESULTS

The calculated friction factors obtained from this investigation, for both vertical and horizontal flow and for both air and water, and with flow in both directions in the test section, are presented in Tables II - XI. The results are also shown graphically in Figures 6-8 as plots of the logarithm of the Reynolds number versus the logarithm of the friction factor, together with the established friction factor relationships for flow in smooth wetted circular metallic and glass tubes. The results obtained by Atallah are also sketched in for comparison.

In general the results appear to be quite consistent and reasonable. It can be seen that in all cases, especially the 3/8" and 3/4" tubes, the friction factors for flow in opposite directions differ. For example, while the results for flow with end 2 of the 3/4" tube at the inlet give friction factors about 15% higher than the established values for wetted flow, the results with the tube reversed give friction factors about 10% lower than those established for wetted flow. The reason for this is undoubtedly that the pressure tap holes are not perfect; fine pressure tap holes, if not drilled normal to the flow or if a drilling burr remains, may result in some impact head being measured, or in separation occurring. The difficulty in drilling the holes is due to the "buttery" nature of Teflon.

It may be observed that the friction factors for the 3/8" nominal I.D. Teflon tube are somewhat higher than the results, at the same Reynolds number, for the other tubes. Koo (Ref. 7) observed that tubes of small diameter (less than 2") show a slight effect of diameter on friction factor, although he considered the effect too small to correlate in his equation. It may also be noted that the slope of the lines which could be drawn through the friction factors for a given diameter tube in the turbulent regime are generally steeper than the slope of the empirical equation for smooth wetted glass and metallic tubes. Koo also points this out, explaining that his correlation holds only within about plus-or-minus 5% and most experimental data which he used fell within that range despite the slightly different slopes.

Although extensive data for horizontal flow of water were obtained only in the 3/4" Teflon tube, with end 1 at the inlet, the remarkable agreement between this data and the corresponding data for vertical flow indicates that there is no noticeable difference between vertical and horizontal flow of water in non-wetted Teflon tubes, at least in the turbulent regime. Although this conclusion is experimentally justified only in the turbulent regime, it seems likely that it will apply in laminar flow also.

The calculated friction factor results in the laminar and transition regions (Reynolds number less than about 3000), can be seen to be considerably more scattered than the turbulent results. There are several reasons for this scattering. First, the pressure drops measured for most of the points in this region were relatively small, of the order of 1/2" on a carbon tetrachloride under water manometer. Because the percent error in measuring such a small pressure drop can be considerable, no points obtained at manometer readings less than 1/2" were included in the graphical presentation, although they are included in the tabulated results for completeness.

A second reason for the scattering of points in the transition region is the fact that most investigators have found flow in this region quite unstable. Some scattering is therefore to be expected.

Finally, there is some doubt concerning the adequacy of the calming lengths employed in this investigation, especially for laminar flow. Nikuradse (Ref. 10) has shown that an inlet length of 25 - 40 diameters is sufficient for a fully developed velocity profile in turbulent flow, which criterion was met in this investigation. In the case of laminar flow, however, the required calming length expressed in pipe diameters is 0.065 times the Reynolds number (Ref. 7). Thus at a Reynolds number of 1000, a calming length of 65 diameters would be required. At higher Reynolds numbers, in the laminar regime, correspondingly longer calming lengths would be required. The possible inadequacy of the calming lengths in this investigation could result in some turbulence in the test section, which would lead to high friction factors in the laminar regime.

It can be seen from the graphical presentation of results that results of this investigation for flow in one direction often approach the results obtained by Atallah. This leads to the conclusion that Atallah's results were incorrect because his experimental procedure was incorrect in that he did not make provision for testing the tubes with flow in opposite directions. Since the tubes had all been somewhat altered since Atallah's investigation, no data agreeing with his results exactly could be obtained.

Because there does not seem to be any difference between friction factors in wetted and non-wetted flow in Teflon tubes, it is probable that there is no basic difference between the flow mechanism in wetted and non-wetted flow. This implies no slip at the wall, although slip might be expected from the nature of a non-wetted surface, which is confirmed by several other investigations (Ref. 4,11,12, 13).

As a consequence of these results, it appears that entrained air bubbles, mentioned before, do not affect friction factors noticeably. Although vortexing in the constant head tank, a condition which tended to support the entrained air bubble theory in Atallah's work, was eliminated in this investigation, it is unlikely that entrained air bubbles were important in Atallah's investigation either.

The limited data obtained for flow of air through Teflon tubes agree within experimental error with the experimental results with water. Since the air flow cannot be considered as either non-wetted or wetted flow, and since the results obtained are in agreement with established equations for friction factors in smooth wetted glass and metallic tubes, this strengthens the conclusion that there is no signigicant difference between wetted and non-wetted flow in Teflon tubes.

With regard to heat transfer in non-wetted flow, it has been proposed by Johnson, et al., (Ref. 5,6) that the low heat transfer rates observed in non-wetted heat transfer to liquid metals may be due to distortion of the velocity profile, presumably due to slip at the wall. Since a distortion of the velocity profile would also change the observed friction factors in non-wetted flow, the results of this investigation indicate that this proposal is improbable.

It has also been mentioned that the existence of a free surface layer of liquid with properties different from the bulk of the liquid may result in a resistance to heat transfer. Since the velocity profile near the wall is determined by the wall shear stress and the fluid viscosity at the wall according to the Newtonian definition of viscosity,

$$\tau_w \, g_c = \mu \frac{dV}{dy}$$

where $\tau_w$ = wall shear stress

$g_c$ = gravitational constant, acceleration due to gravity

$\mu$ = fluid viscosity

$\frac{dV}{dy}$ = velocity gradient

it can be seen that an increase in fluid viscosity at the wall, with wall shear stress constant, will result in a decrease in the velocity gradient near the wall, i.e., a thicker laminar layer. It is quite generally believed that the viscosity near the free surface may be many times the viscosity in the bulk of the liquid due to the strong cohesive forces in the surface fluid resulting from unsatisfied surface forces. The result of this thicker laminar layer will be higher fluid velocity in the center of the tube because the tube will be effectively decreased in diameter. Also, the thicker laminar layer will result in a larger resistance to heat transfer, especially in a fluid with poor molecular diffusivity of heat such as water. This proposal does not appear to be very promising in the light of the very limited data available. First, the forces involved in the free surface at a non-wetted interface are of extremely short range, of the order of one molecular diameter. A monomolecular layer of liquid with different properties at the surface could not account for the discrepancies observed in non-wetted heat transfer. Secondly, the experimental evidence indicates higher resistance in heat transfer to liquid metals in non-wetted flow. The thicker laminar layer would not be expected to have very much effect on fluids with high molecular diffusivities of heat such as liquid metals. A third failure of this proposal is that it does not account for the electrical resistance which has been observed at the non-wetted interface, unless it is assumed that the electrical conductivity of the fluid at the surface is increased.

Several investigators have suggested gas bubble entrainment or an insulating layer of gas at the tube wall as an explanation for low heat transfer results under non-wetted conditions. Johnson, et al., (Ref. 5) discredited this explanation on the basis of tests in which they measured heat transfer coefficients to mercury in an atmosphere of first helium and then argon, with 1/9 the thermal conductivity of helium. If the gas film were important, considerably different results should be obtained. In very careful tests, however, they observed that the differences were less than a few percent.

Finally, an oxide film at the solid - liquid interface has been suggested. Johnson, et al., (Ref. 6) discredited this idea on the basis that tests with water, in the same apparatus used for the liquid metal tests, gave good results while the liquid metal results were low. It has been mentioned earlier, however, that a slight additional thermal resistance may not be significant in water heat transfer, because of its high Prandtl number while the effect in liquid metals could be very significant. The required oxide film thickness is very small, of the order of a few thousandths of an inch (Ref. 2), and the occurrence of wetting in liquid metal systems may involve removal of this oxide film, thus leading to the conclusion that the wetting itself increased heat transfer. This explanation might also account for the frequently erratic behavior of liquid metal systems, since the oxide film may not always be removed when wetting is observed.

A phenomenon which has attracted some interest recently is the apparent interfacial resistance in mass transfer. It is possible that a similar phenomenon occurs in non-wetted heat transfer, possibly due to a rearrangement of molecules at the non-wetted interface where the free energy is high and unbalanced forces are present. This free energy is dissipated as heat of wetting when wetting occurs. This type of molecular rearrangement could also explain the electrical resistance which has been measured at the non-wetted interface in liquid metal systems. This may indicate a region of small thickness and very high resistance.

While there is no data to support the oxide film theory, it seems capable of explaining most of the disagreement observed in non-wetted heat transfer. It is very likely, however, that all of the factors which have been considered play some part, more or less, in the low heat transfer coefficients observed in liquid metal heat transfer. The possibility of the existence of a region of small thickness and very high resistance is particularly interesting, but no method is available at present to substantiate this suggestion by experiment.

## CONCLUSIONS

1. The results of this investigation indicate that there is no significant difference between friction factors for horizontal or vertical flow of water in non-wetted Teflon tubes.

2. The friction factors measured for non-wetted flow agree quite well with published friction factors for wetted flow in smooth circular tubes.

3. Limited results with air flow in Teflon tubes give approximately the same friction factors as for water flow.

4. Atallah's results were in error due to incorrect experimental procedure.

# FUTURE WORK

An investigation of heat transfer through Teflon tubes is planned, using the apparatus described in this report. It is expected that inside film coefficients and condensing film coefficients will be obtained, and it is hoped that some insight into the difference between heat transfer in wetted and non-wetted flow may be obtained.

APPENDIX

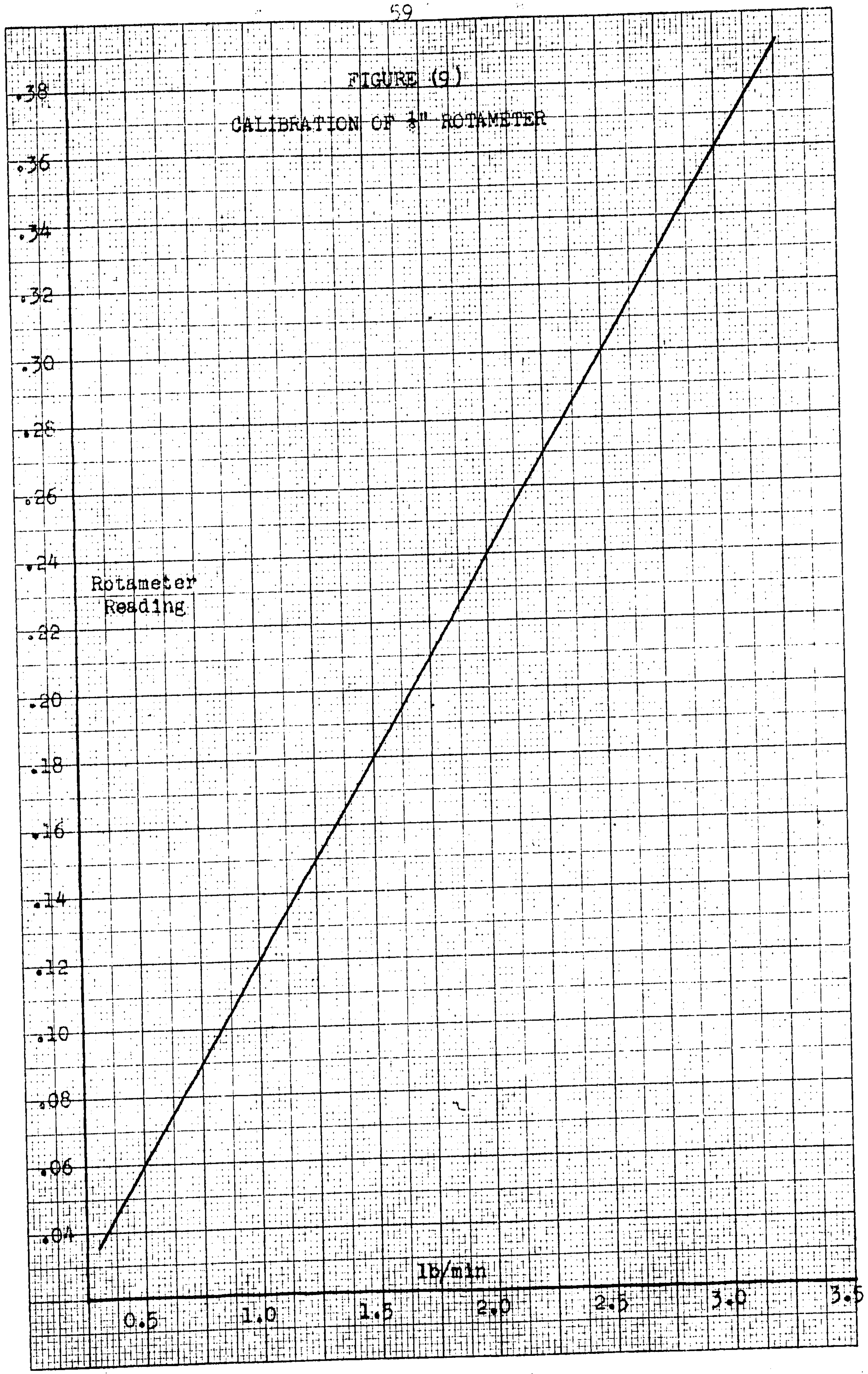


FIGURE (9)
CALIBRATION OF ½" ROTAMETER
Rotameter Reading
lb/min
.38
.36
.34
.32
.30
.28
.26
.24
.22
.20
.18
.16
.14
.12
.10
.08
.06
.04
0.5
1.0
1.5
2.0
2.5
3.0
3.5

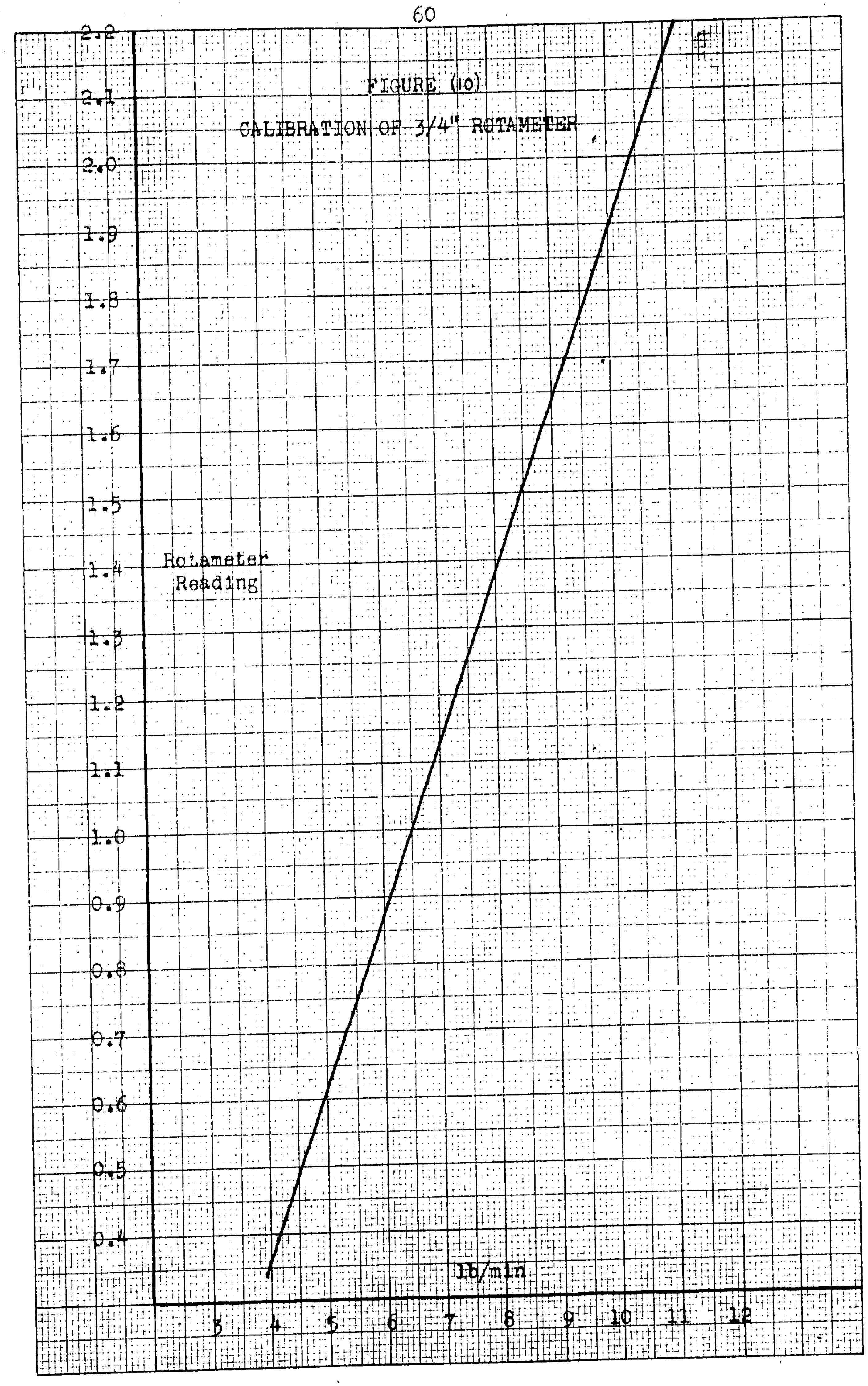
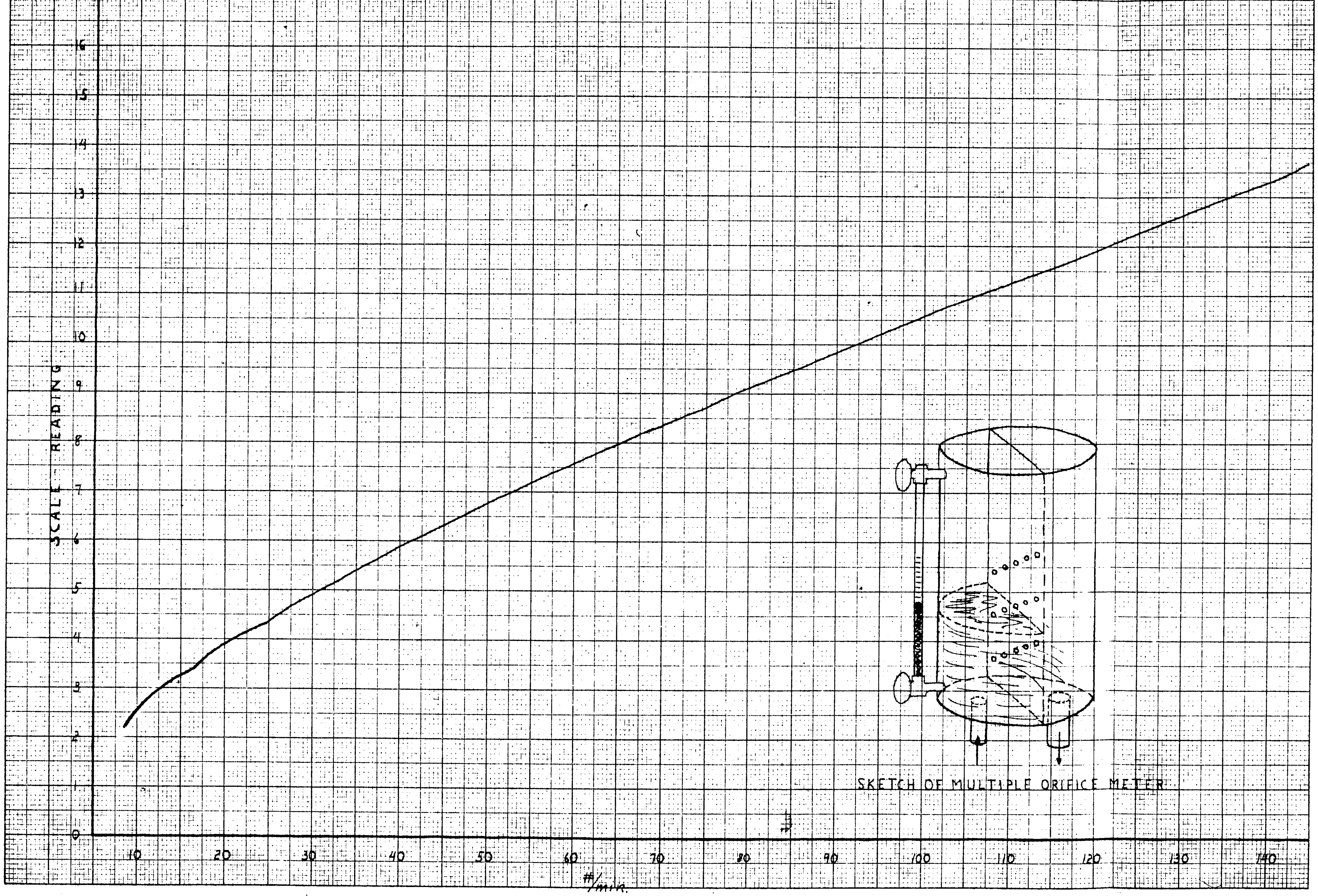


FIGURE (10)
CALIBRATION OF 3/4" ROTAMETER
Rotameter Reading
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
lb/min
3
4
5
6
7
8
9
10
11
12

FIGURE (11)
CALIBRATION OF MULTIPLE ORIFICE METER



## SAMPLE CALCULATIONS

The following is a sample calculation for Run # 129, vertical water flow in 3/4" Teflon tube, end 2 at inlet.

Experimental Data:

| | |
|---|---|
| Water Temperature | $49^{\circ}$ F |
| Multiple Orifice Meter | 7.04 |
| Pressure Drop, inches carbon tetrachloride under water | 10.88 |

### Calculation of Reynolds Number

$$Re = \frac{DV\rho}{\mu}$$

D = tube diameter, .721 inches

V = linear fluid velocity,

$$= \frac{(52.70 \text{ lbs./min.})}{(62.43 \text{ lbs./cu.ft.})(60 \text{ sec./min.})(.00283 \text{ ft.}^2)}$$

= 4.97 feet per second

$\rho$ = 62.43 lbs./cu. ft. at $49^{\circ}$ F

$\mu$ = 1.3462 centipoises at $49^{\circ}$ F

= 1.3462cp x .672#/ft. sec./ cp x $10^{-8}$

$$Re = \frac{0.721/12 \times 4.97 \times 62.43}{1.3462 \times .000672} = 20600$$

### Calculation of Friction Factor

$$f = \frac{\pi^2 \rho\, g_c D^5 \Delta P}{32\, L\, w}$$

$$\Delta P = \frac{10.88''}{12}(1.595-1.00) \times 62.43 = 33.70 \text{lb.force/ft}^2$$

$g_c$ = 32.174

w = 52.70/60 = .878 lbs./sec.

L = 32 inches/12 = 2.67 feet

$$f = \frac{(3.14) \times 62.43 \times 32.174 \times (.06) \times 33.70}{32 \times 2.67 \times .878}$$

= .00795

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